

10/584759

Docket No.: WMB-12405


AP20 Rec'd PCT/PTO 27 JUN 2006

CERTIFICATION

I, the below named translator, hereby declare that: my name and post office address are as stated below; that I am knowledgeable in the English and German languages, and that I believe that the attached text is a true and complete translation of PCT/AT2004/000405, filed with the Austrian Patent Office on November 18, 2004.

I hereby declare that all statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true; and further that these statements were made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code and that such willful false statements may jeopardize the validity of the application or any patent issued thereon.

Hollywood, Florida

  
Rebekka Pierre

June 27, 2006

Lerner Greenberg Sterner LLP  
P.O. Box 2480  
Hollywood, FL 33022-2480  
Tel.: (954) 925-1100  
Fax.: (954) 925-1101

**METHOD AND DEVICE FOR CONVERTING HEAT INTO MECHANICAL WORK**

**BACKGROUND OF THE INVENTION**

The invention relates to a method for converting heat into mechanical work, in which a working medium is compressed in a cyclic process while giving off heat and is subsequently brought in thermal contact with the ambient environment via a first heat exchanger, is then expanded while obtaining mechanical work, whereupon the cyclic process is run through again.

**DESCRIPTION OF THE PRIOR ART**

Numerous working methods are known to convert thermal energy into mechanical work. Usually, a working medium is compressed, heated, expanded in the heated state and cooled in such cyclic processes, whereupon the cyclic process starts again. The precondition for such cyclic processes is that two different temperature levels are available which are used for heating or cooling the working medium. Generally, a certain temperature is defined as the ambient temperature, which is the temperature of a medium which is available in an unlimited and gratuitous way. This can be the air temperature of the ambient environment for example or the temperature of a water body from which water can be taken in sufficient quantities for purposes of temperature exchange.

No cyclic processes are known with which it is possible to gain mechanical work from thermal energy without disposing over a heat transfer medium whose temperature differs substantially from ambient temperature. According to current

1 substantially from ambient temperature. According to current  
2 belief such a cyclic process is excluded by the second law of  
3 thermodynamics. It is stated in a more precise version of the  
4 second law of thermodynamics that the efficiency of any  
5 cyclic process for converting thermal energy into mechanical  
6 work cannot exceed the so-called Carnot efficiency which is  
7 calculated from the ratio of the available temperature  
8 levels. Real existing methods and apparatuses are generally  
9 also far away from the Carnot efficiency.

10  
11 Apparatuses for generating temperature differences are known  
12 which use gas-dynamic effects occurring at high accelerations  
13 in order to produce temperature differences. These  
14 apparatuses are not suitable in order to perform cyclic  
15 processes for gaining mechanical work.

16  
17 DE 38 12 928 A shows an apparatus which tries to overcome the  
18 above disadvantages. Even with such an apparatus it is not  
19 possible to improve the efficiency to a relevant extent.

20  
21 It is the object of the present invention to provide a method  
22 of the kind mentioned above which allows obtaining mechanical  
23 work from thermal energy with the highest possible  
24 efficiency.

25  
26 It is a further object of the invention to provide an  
27 apparatus with which the method in accordance with the  
28 invention can be performed.

29  
30 SUMMARY OF THE INVENTION

31  
32 In accordance with the invention, this method is  
33 characterized in that the working medium, after expansion, is  
34 guided through another heat exchanger which is situated

1 inside a rapidly rotating rotor and which, on the exterior  
2 thereof, is surrounded by at least one essentially annular  
3 gas chamber from whose exterior heat is dissipated.

4  
5 The inventor of the present invention has recognized that by  
6 including the static gas theory in connection with  
7 considering gravity or acceleration acting upon the gas  
8 molecules or atoms it is possible to illustrate cyclic  
9 processes which have an especially high efficiency. The  
10 problematic aspect in this connection is however that the  
11 effects caused by gravity are very small, as a result of  
12 which technical implementation is very difficult. As a result  
13 of the cyclic process in accordance with the invention, the  
14 utilization of thermal energy for generating mechanical work  
15 can be achieved under economically viable framework  
16 conditions. A substantial precondition for the method in  
17 accordance with the invention is the achievement of the  
18 highest accelerations by a rapidly running rotor, with the  
19 achieved acceleration values being chosen as high as  
20 possible.

21  
22 It is especially preferable when the working medium is guided  
23 downstream of the rotor through a compressor. The heating  
24 caused in the compressor is so low in any case that the  
25 working medium cooled in the rotor remains beneath the  
26 ambient temperature. This ensures that the working medium  
27 takes up ambient heat in the first heat exchanger.

28  
29 In an especially advantageous embodiment of the method in  
30 accordance with the invention it is provided that the working  
31 medium is guided in the axial direction through the rotor.  
32 The effects of high acceleration in the interior of the rotor  
33 on the pressure conditions in the working medium can be  
34 eliminated substantially.

1  
2 The present invention further relates to an apparatus for  
3 withdrawing heat at ambient temperatures with a rotor having  
4 a heat exchanger which can be flowed through substantially in  
5 the axial direction and which is delimited on its outside by  
6 a cylindrical wall on the outside of which there is provided  
7 at least one substantially annular gas chamber.

8  
9 This apparatus is characterized in accordance with the  
10 invention in such a way that the heat exchanger is provided  
11 with a substantially ring-cylindrical configuration and that  
12 the gas chamber is subdivided in the radial direction into  
13 several ring-cylindrical partial chambers. These partial  
14 chambers can have the same dimensioning in the radial  
15 direction, but can also be provided with different  
16 configurations. Only the described configuration of the rotor  
17 allows realizing a cyclic process of the kind mentioned above  
18 in a technically and economically viable manner.

19  
20 It is principally possible that in each of the individual  
21 partial chambers the same gas is present. In such a case, the  
22 pressure on the outside of a partial chamber is generally  
23 higher than the pressure on the inner side of the further  
24 partial chamber adjacent to said partial chamber. This means  
25 that although the pressure increases from the inside to the  
26 outside as a result of centripetal acceleration within the  
27 individual partial chambers, this increase is interrupted at  
28 the boundaries of the individual partial chambers. This leads  
29 to a mechanical loading of the separating walls between the  
30 individual partial chambers. This is technically controllable  
31 because the resulting pressure force acts towards the outside  
32 and the separating walls are not loaded to bulging.  
33 Preferably however, different gases are received in the  
34 different partial chambers which especially have different

1 critical temperatures and pressures. It can thus be achieved  
2 that the pressure load of the separating walls is minimized  
3 because in the balanced state substantially the same pressure  
4 applies inside and outside. It is also within the scope of  
5 the present invention that gas mixtures are used instead of  
6 pure gases, which gas mixtures form concentration gradients  
7 during the operation of the apparatus.

8  
9 As a result of the extremely rapid rotation of the rotor, the  
10 pressure present in the interior of the rotor differ in the  
11 idle state substantially from those in the operating state.  
12 In order to minimize the loading of the separating walls and  
13 the other components, it is provided in an especially  
14 preferred embodiment of the invention that a pressure control  
15 device is provided which is in connection with the ring-  
16 cylindrical partial chambers in order to set the internal  
17 pressure. In an especially preferred manner, the ring-  
18 cylindrical partial chambers are separated from each other by  
19 thin-walled cylindrical separating walls. Mechanical loads of  
20 the individual components can thus be minimized.

21  
22 BRIEF DESCRIPTION OF THE DRAWINGS  
23

24 The present invention will be explained below in closer  
25 detail by reference to the embodiments shown in the drawings,  
26 wherein:

27  
28 FIG. 1 shows a schematic view of an apparatus for performing  
29 the method in accordance with the invention;

30  
31 FIG. 2 shows a rotor of the apparatus of FIG. 1 on an  
32 enlarged scale;

33  
34 FIG. 3 shows a sectional view along line III-III in FIG. 2;

FIG. 4 shows a diagram illustrating the temperature curve in the radial direction of the rotor, and

FIG. 5 shows a Ts-diagram explaining the cyclic process.

#### DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The apparatus of FIG. 1 consists of a turbine 11 for the expansion of the working medium, which turbine is divided into two sections 11a, 11b. A heat exchanger 11c is provided in the first section 11a in order to enable an isothermal expansion. It is principally possible to provide several turbine stages in which the working medium is expanded in an adiabatic way and the heat exchangers are provided between the turbine stages, as a result of which only an approximately isothermal expansion is achieved. If the heat exchanger 11c and the turbine 11 are provided themselves, then it is actually possible to achieve a substantial isothermal expansion. The adiabatic expansion occurs in the second section 11b of the turbine 11. The cooling medium is therefore present at the output of the turbine 11 with a temperature which lies beneath the ambient temperature.

A generator 12 is driven by the turbine 11 and a rotor 13 of a centrifuge is simultaneously driven which is flowed through by the working medium in the axial direction. Compression occurs in a turbine 14, whereupon the working medium is guided back to the turbine 11 again via a recirculation line 15.

Rotor 13 comprises a ring-cylindrical heat exchanger 18 and several gas chambers 17a, 17b, 17c, 17d which are also provided with a ring-cylindrical configuration and lie

1 outside of the heat exchanger 18. Notice must be taken that  
2 the dimensions of the heat exchanger 18 and the gas chambers  
3 17a, 17b, 17c, 17d in the radial direction are shown on an  
4 excessive scale in FIG. 1, because in the case of real  
5 configurations these dimensions are very small and the heat  
6 exchanger 18 and the gas chambers 17a, 17b, 17c, 17d lie  
7 close to the outer jacket of the rotor 13. Rotor 13 is  
8 provided on its outer side with cooling ribs 19 which  
9 represent a heat exchanger for dissipating heat. This is  
10 indicated by arrows 20.

11  
12 The gas chambers 17a, 17b, 17c, 17d are preferably filled  
13 with different gases, with the innermost gas chamber 17a  
14 being filled with helium for example, the adjacent gas  
15 chamber 17b with xenon, the third gas chamber 17c with  
16 nitrogen or a suitable hydrocarbon, and the outermost gas  
17 chamber 17d with a suitable coolant. As a result of the rapid  
18 rotation of the rotor 13, a temperature drop from the outside  
19 to the inside is caused in the gas chambers 17a, 17b, 17c,  
20 17d which strongly cools the working medium in the heat  
21 exchanger 17.

22  
23 Heat at the temperature level of the ambient environment is  
24 supplied to the heat exchanger 16, which is indicated by  
25 arrow 21. An increase in the efficiency can be achieved when  
26 the waste heat of the rotor 13 is also supplied to the heat  
27 exchanger 16 according to arrows 20.

28  
29 FIG. 2 shows the rotor 13 in detail in a modified embodiment.  
30 The working medium is supplied in the interior of a hollow-  
31 bored first shaft 22 which is held in a bearing 23 and is  
32 guided via distributor lines 24 to the outside radially to  
33 the heat exchanger 18. In the interior of the heat exchanger  
34 18 the working medium flows in the axial direction to the



1 opposite side of rotor 13 in order to be guided in further  
2 distributor lines 25 radially to the inside to a further  
3 shaft 26 held in a bearing 27. As in the preceding  
4 embodiment, the four gas chambers 17a, 17b, 17c, 17d are  
5 provided radially inside one another. A heat exchanger 18 is  
6 provided on the outside for dissipating heat. A housing 28 is  
7 indicated in a schematic manner, in which the rotor is  
8 arranged in a rotatable way which comprises a plurality of  
9 magnets 29. The magnets 29 are used for relieving the  
10 bearings 23 and 27 at high speeds and are in interaction with  
11 magnets (not shown) on the outside of rotor 13 itself. The  
12 polarity is directed in such a way that the magnets 29 and  
13 the magnets on rotor 13 repulse one another, as a result of  
14 which an inwardly facing force is exerted on the jacket  
15 surface of the rotor 13 which considerably reduces mechanical  
16 stress as a result of centrifugal forces and allows higher  
17 speeds. At least one gas container 30 is provided in the  
18 interior of the rotor 13, which gas container is in  
19 connection with one of the gas chambers 17a, 17b, 17c, 17d  
20 via lines (not shown). Preferably however, the compensating  
21 reservoir 30 comprises sub-containers (not shown) which are  
22 individually connected with the individual gas chambers 17a,  
23 17b, 17c, 17d. The mean pressure level in the gas chambers  
24 17a, 17b, 17c, 17d can thus kept at a predetermined value  
25 irrespective of the respective speed of the rotor 13, so that  
26 mechanical loading of the separating walls between heat  
27 exchanger 18 and the gas chambers 17a, 17b, 17c, 17d remains  
28 within predetermined limits.

29  
30 The following tables 1 to 4 show by way of an embodiment the  
31 state variables of the gases or gas in the individual gas  
32 chambers 17a, 17b, 17c, 17d, with table 1 relating to the  
33 innermost gas chamber 17a, table 2 to the gas chamber 17b,  
34 table 3 to the gas chamber 17c and table 4 to the gas chamber

17d. The left half of the table indicates the state variables on the outside wall of the respective gas chamber 17a, 17b, 17c, 17d and the right half of the table indicates the respective state variables on the inner wall of the respective gas chamber 17a, 17b, 17c, 17d.

The references mean the following in the tables 1 to 4:

T            Temperature in K  
d            Density in kg/m<sup>3</sup>  
p            Pressure in MPa  
s            Entropy in kJ/kgK  
u            Inner energy in kJ/kg  
h            Enthalpy in kJ/kg

Table 1

T	276.32	T	121.51
d	174.43	d	28.62
p	14.33	p	0.91
s	5.18	s	5.18
u	173.15	u	81.95
h	255.33	h	114.07

Table 2

T	424.17	T	276.32
d	129.39	d	50.25
p	17.61	p	4.07
s	5.62	s	5.62
u	294.47	u	195.45
h	430.58	h	276.45

Table 3

T	579.04		T	424.17
d	94.29		d	45.76
p	17.54		p	5.88
s	5.98		s	5.98
u	419.52		u	307.62
h	605.58		h	436.27

1

2 Table 4

T	739.98		T	579.04
d	77.64		d	42.67
p	18.39		p	7.54
s	6.24		s	6.24
u	550.60		u	426.66
h	787.48		h	604.32

3

4

5 FIG. 3 schematically shows a sectional view along line III-  
6 III in FIG. 2, with the heat exchanger 18 and the cooling  
7 ribs 19 having been omitted for improving clarity of the  
8 illustration. Arrows 20 symbolize the heat flow.

9

10 FIG. 4 shows a diagram which schematically states the  
11 temperature distribution in the radial direction of the rotor  
12 13 which is stated by  $r$ . The curve  $K_1$  represents the  
13 temperature  $T$  in the idle state, i.e. when no heat flow  
14 occurs, which is the case when the rotor 13 is insulated on  
15 the inside and outside. Curve  $K_2$  represents the temperature  $T$   
16 in operation, i.e. when there is a heat flow in the radial  
17 direction.

18

19 FIG. 5 shows an idealized  $T/s$  diagram, in which the  
20 temperature is entered over the entropy. The cyclic process  
21 is passed in the direction of the arrows 31. The double arrow  
22 32 shows the temperature difference of the centrifuge, i.e.

the rotor 13 over the gas chambers 17a, 17b, 17c, 17d. As a result of the losses in the heat transfer, the temperature difference 33 which can actually be used in the cyclic process is considerably lower. The states 1, 2, 3, 4 in the diagram correspond to the states in the analogously designated points in FIG. 1. It can be noted for example that in the case of a single-phase working medium the changes in state 1 -> 2 and 3 -> 4 are not precisely isothermal.

Table 5 indicates the state variables in the individual points under idealized assumptions.

	[K]	[kg/m <sup>3</sup> ]	[MPa]	[kJ/kgK ]	kJ/kg	kJ/kg
	T	d	P	s	u	h
Point 1	130	15	0.54937 258	5.44088 686	92.1986 033	128.8234 42
Point 2	130	70	2.10257 662	4.92707 388	77.8876 766	107.9244 86
Point 3	283	316.50 07	30.2486 572	4.92707 388	153.810 311	249.3824 76
Point 4	283	92.150 807	7.66041 346	5.44088 686	192.911 843	276.0409 41

The present invention allows realizing an apparatus and a cyclic process with efficiencies which substantially exceed those of conventional solutions.